Journal of Achievements in Materials and Manufacturing Engineering
VOLUME 20
ISSUES 1-2
January-February
2007

Diskrete-cotinuum methods application for rotating machine-absorber interaction analysis

Z. Stocko*, B. Diveyev, V. Topilnyckyj
Lviv National Polytechnic University, St.Bandera str., 12, Lviv, Ukraine
* Corresponding author: E-mail address: stotsko@polynet.lviv.ua

Received 03.11.2006; accepted in revised form 15.11.2006

Analysis and modelling

ABSTRACT

Purpose: The main aim of this paper is improved dynamic vibration absorbers design with taking into account complex rotating machines dynamic.
Design/methodology/approach: The numerical schemes row is considered for the complex vibroexitated constructions. Methods of decomposition and the numerical schemes synthesis are considered on the basis of new methods of modal synthesis.
Findings: Development of mathematical models of complicated machines and buildings in view of their interaction with system of dynamic vibration absorbers. Dynamic vibration absorbers – complicated rotating machines system design optimized on vibro- absorption properties.
Research limitations/implications: The research must be done for non-linear rotor dynamics.
Practical implications: The absorbers designed in accordance with this paper can be applied not only to electric machines or aeronautic structures, but to any other type of vibro-exitated structure, such as cars, chisel installation, optical, magneto-optical disks, washing machine, refrigerator, vacuum cleaner, etc.
Originality/value: The paper has novelty both in theoretical, and in practical aspect, In order that optimal parameters of DVA be determinate the complete modeling of dynamics of rotating machine should be made. Traditional design methodology, based on discontinuous models of structures and machines are not effective for high frequency vibration.
Keywords: Engineering design; Rotating machines; Discreet-continuum theory; Modal analysis; Dynamic vibration absorber

1. Introduction

The most effective way to solve the problem of vibration decreasing is to apply optimally designed dynamic absorber or a set of such absorbers. It is desirable to eliminate unwanted vibration in many applications. On of the most visible applications is transportation. In automobiles, aircraft, and watercraft vibration can cause irritation and even motion sickness to the occupants, and can also cause accelerated wear and mechanical fatigue to the vehicle hardware. Other applications include manufacturing equipment, where imbalances of rotating machinery can cause eccentric rotation, which can degrade surface finishing and machining tolerances. As the speeds of electronic information storage devices, such as compact discs, DVDs, and hard drives increase, the effect of imbalance on the rotating discs becomes increasingly important. Still it is possible to give another dozen of examples of machines and constructions where DVA application is expedient.

The main aim of the paper is the improved dynamic vibration absorbers (DVA) design with taking into account complex rotating machines dynamic. It is often impossible to balance the rotating elements to reduce the vibration to an acceptable level. The paper contemplates the provision of DVA or number of such DVA. Such originally designed DVA reduces vibration selectively in maximum mode of vibration without introducing vibration in other modes. The final results are achieved at far less expense than would be required to replace the concrete and steel foundation with one massive enough. By installing DVA, one can minimize excitation virtually at the source. In order to be more effective, a vibration absorption system should react in all frequency domains. The present absorber also has as an advantage that it can be constructed such that it has a wide-range vibration absorption property. This construction allows for the easy connection of above the rotor equipment.

2. Methodology of machine and absorber modeling

In order optimal parameters of DVA to be determinate the complete modeling of dynamics of rotating machine is obvious. The two degrees of freedom model is totally inadequate to calculate the vibration frequencies of the construction with accuracy and therefore, for a sufficiently accurate determination of its dimensional characteristics so as to determine such frequencies. It is therefore necessary in practice to dimension the construction through more complex modeling. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination.

2.1. Rotating machine modeling

Large rotating elements, particularly such elements as exhaust fan rotors used in electric power generating plants or in gas compression, are unbalanced in operation due to their exposure to varying factors. It is often impossible to balance the rotating elements to reduce the vibration to an acceptable level. The two degrees of freedom model described in [1-3] is totally inadequate to calculate the natural frequencies of vibration of the rotor with accuracy and therefore, for a sufficiently accurate determination of its dimensional characteristics so as to determine such frequencies. It is therefore necessary in practice to dimension the rotor through more complex modeling [4-6]. In particular, concentrated mass and rigidity calculation methods may be adopted based on an even more accurate theoretical determination.

The numerical schemes (NS) row is considered for the complex vibro-exitated constructions. Methods of decomposition and the NS synthesis are considered on the basis of the new methods of modal synthesis. Complex NS are led of discretely-continuum type that enables in the adaptive mode to calculate tension not only in the continue elements, but in places of most tension concentration in joints. The absorbers in accordance with this project may be applied not only to electric machines ore aeronautic structures, but also to any other type of vibro-exitated structure, such as cars, chisel installation, optical, magneto-optical disks, washing machines, refrigerators, vacuum cleaners, etc. Rotating machinery will typically introduce both acoustic and vibration energy into any fluids or structures surrounding the machinery. Both random and deterministic processes related to the operation of the machinery can cause the acoustic and vibration energy. Random processes result in noise or vibration that is spread over a wide band of frequencies. Deterministic processes, on the other hand, often generate energy that is confined to a family of distinct frequencies radiated as "pure" tones. Methods of decomposition and the NS synthesis are considered. on the basis of new methods of modal synthesis [7-10].

In fig. 1 the typical discretely-continual scheme rotating machine is presented, namely - the compressor (though this scheme is general for many rotating machines), with 5 continual elements - rotors R_1, R_2, the case K, the base Φ, pipes T, and discrete ones: bearings K_{ij}, a clutch K_M, elastic joints of pipes to the case and the base of the compressor and elastic joints of case to the base(though there maybe several points of a compression, they are schematically represented in fig. 1 by elements: K_1, K_η^K and K_Φ). Let's take that vibration spatial kinematics loading on bearings of rotors (it will be defined on the basis of the complex calculation scheme from fig. 1) is given. We shall consider the following reduced NS: U_{ij}^0 - The dynamic displacement affixed on the part of the case as vectorial magnitudes (rolling contact bearings of such type are considered, that it is possible to disdain longitudinal gains and the moments)

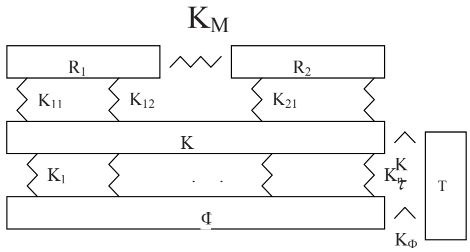

Fig. 1. The scheme of rotating machine

Let's consider dynamics of the rotating machine shaft taking into account flexibility of the shaft both on curving and on torsion. We shall proceed from a variation principle of Hamilton. We shall accept kinematics hypotheses for deviations of an axis from a certain statically counterbalanced position of the shaft on bearings. We suppose that the center of masses of the given cross-section of the shaft coincides with its center of weight in the immovable frame rigidly connected to statically equilibrium position of the shaft on elastic support. Then the kinetic energy will consist of two items

$$T = T_w + T_{rot}, \tag{1}$$

T_w – bending component, T_{rot} – rotation one. The first item is equal to

$$2T_w = \int_0^L \rho\, V_C^2 dz = \int_0^L \rho\, (V_r^2 + V_\varphi^2) dz, \tag{2}$$

$\rho(z)$ - a linear mass of a shaft, V_C^2 – quadrate of tangential velocity of shaft equal the sum of quadrates of radial V_r and tangential $V_{\varphi r}$ velocities of the center of the given cross-section of the shaft, multiplied on its mass of relative movement will be equal to

$$2T_{rot} = \int_0^L \rho R^2 \gamma_\varphi^2 dz \ . \tag{3}$$

Here $R(x)$ - radius of the given part of the shaft, γ_φ - axial angular velocity. Here gyroscopic composites [1-3] are neglected.

Let's accept kinematics hypotheses for deformation of the shaft. We assume, that in an immovable frame (z, r, φ) , the shaft realizes elastic rotating oscillations with a changeable angle of twisting

$$\varphi = q_\varphi(t)\lambda_\varphi(z), \tag{4}$$

and radial oscillations in a mobile frame (z, r, φ), rigidly connected to shaft in orthogonal planes [1]

$$W_1 = \sum_i q_i^1(t)\lambda_i(z), \qquad W_2 = \sum_i q_i^2(t)\lambda_i(z) \ . \tag{5}$$

Here $q_i^j(t)-$ are time dependent unknown functions and multipliers by its $\lambda_i(z)-$ known coordinate functions.

Let's note kinetic and potential energy. For the last we assume resistance of the shaft to be linearly elastic. According to (1) - (5) we obtain [7-10]

$$(M^c \ddot{q}^c + \overline{K}^c \cdot q^c)\cdot \delta q^c + \sum (M_i q_i^n)\delta q_i^n = 0 \ . \tag{6}$$

Here M^c – the mass matrix of continua elements, M_i – the discrete elements masses, δq_i^n – independent time functions variations. By equating the terms by these variations we obtain the system of ordinary differential equations [7-10].

2.2. Dynamic absorber modeling

Some example of beam-like DVA with prescribed damping properties are presented in Fig.2 Consider plain bending of plate in elastic joint (b). The detail of elastic and damping property evaluation may be found in [11,12]. Material of plate is assumed more rigid than that of interlayer (fig.3). Material of plate is anisotropic, with elasticity modules $\boldsymbol{C_{xx}}$, $\boldsymbol{C_{xz}}$, $\boldsymbol{C_{zz}}$, $\boldsymbol{G_{xz}}$. Material of interlayer is isotropic and incompressible (rubber-like material). Such joint is widely used in industry. Equilibrium equations are derived on the basis of such kinematical hypothesis [13]

$$u = u_{ij}\cdot x^i \cdot z^{j-1}, \quad w = w_{ij}\cdot x^i \cdot z^{j-1}, \tag{7a}$$

or

$$u = u \cdot x^{i-1}\cdot z^{j-1}, \quad w = w\cdot x^{i-1}\cdot z^{j-1}, \tag{7b}$$

for unclamped left corner of plate. Put (7) to the (8) and get analog of (6)

$$\begin{aligned}
&\int_{V_p} (\sigma_{xx}\delta\varepsilon_{xx} + \sigma_{zz}\delta\varepsilon_{zz} + \tau_{xz}\delta\varepsilon_{xz})dV + \\
&+ \int_0^L (K^+(x)w^+ + K^-(x)w^-\delta w^-)dx + \\
&+ \int_0^L (K_u^+(x)u^+\delta u^+ + K_u^-(x)u^-\cdot\delta u^-)dx - \\
&- \int_{-H_p}^{H_p} (N(z)\delta u + T(z)\delta w)\cdot dz = 0.
\end{aligned} \tag{8}$$

Here $\boldsymbol{K}^{+-}$, $\boldsymbol{K}^{+-}_v$ are respectively normal and tangential elastic coefficient of elastic interlayer us Winkler foundation one. There may be found by the same method [13].

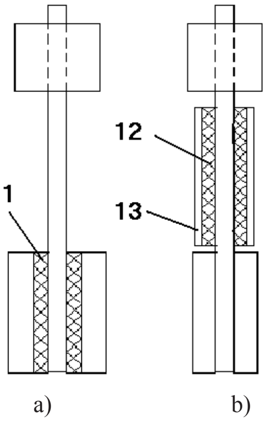

Fig. 2 DVA with: a)damping layer in the joint (11); b) attached constrained (13) damping layer (12)

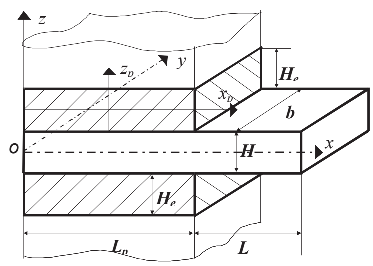

Fig. 3 Elastic joint of beam to base

3. Description of achieved results of own researches

On the Fig.4 sum numerical results are presented of plate elastically clamped to point 5.0

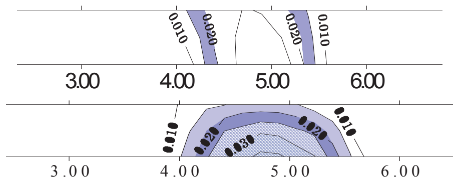

Fig. 4. Normal stresses σ_{zz} for distinct anisotropy level (E/G=0.4; 0.1).

In Fig. 5 is presented the result of rotating machine optimization in frequency domain with DVA. For these purples the linearization of system (6) was provided. The one mass model for rotor and case was considered [1-3]. The DVA was attached to the case.

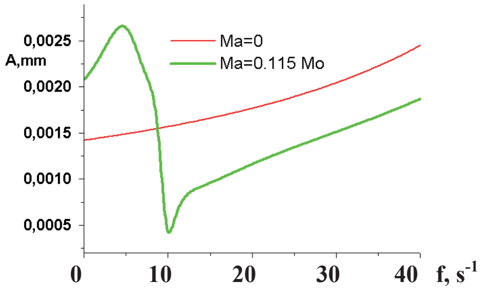

Fig. 5. Dynamic response with and without absorber

Here **Ma** is the mass of absorber and **Mo** – mass of the case. The frequency response lines **Ma=0** and **Ma=0.115Mo** presents the case of vibration: levels **A** without and with DVA with optimal mass. The result may be much improved by changing the special form of the DVA [10].

Forces obtained on the basis of the given calculation schemes may be used further for determine deformed conditions of cases, bases and to define vibrations levels for elements of the rotating machine. Algorithms adduced in [7-10], and known programs – such as ANSYS, NASTRAN, COSMOS may be used by calculation of bearings and shaft stresses at the given above loading.

4. Conclusion

In order that optimal parameters of DVA are determinate the complete modeling of dynamics of rotating machine should be made. Traditional design methodology, based on discontinuous models of structures and machines are not effective for high frequency vibration. They do not give a possibility to determine strains and to predict durability. Present research develops a modern prediction methodology, based on complex discreet-continuum theory. This allows to take into consideration system anisotropy, supporting structure strain effect on equipment motions and to determine some new effects that are not described by ordinary mechanics, namely concentration of strain in junctions and chaotic oscillations.

References

[1] S.P. Timoshenko, Oscillations in engineering, - M., The Science, 1967, 444
[2] A. Tondl, Dynamics of a rotor of turbogenerators, - L., The Energy, 1971, 387.
[3] F.M. Dimentberg, Bending oscillations of rotating shaft. AS the USSR, 1959.
[4] Z. Dabrovsky, I. Komorska, A. Puchalski, Diagnostic of manufacturing and assembling errors of wibrating system, Warshawa-Radom, 2001, 108.
[5] H.D. Nelson, A Finite Rotating Shaft Element Using Timoshenko Beam Theory, ASME J. Mech. Des., 102 (1980) 793 - 803.
[6] T. Zheng, N. Hasebe, Nonlinear Dynamic Behaviors of a Complex Rotor-Bearing System, ASME J Appl. Mech., 67 (2000) 485-495.
[7] B. Diveiev, Algorithms of calculation of the displayed designs in CAD and diagnostic systems on microcomputers. Materials 4 and 5 International scientific - practical conferences, Ukrsoft, Lviv, 1995, 191-194.
[8] P. Hashtchuk, B. Diveyev, I. Vayda, I. Vicovich, Design of girder booms for sprayers of increased durability and high quality of agent application 'Vibration in physical systems', XIX symposium, Abstract and invited lecture, Poznan-Blazejewko, 2000, 44-45.
[9] I. Butiter, B. Diveyev, I. Vikovych, Optimization and design of the girder spray booms of inereased durability and quality of an agent sprayind 'Vibrations in physical systems', XX jubilee sysmposium, Abstract and invited lecture, Poznan-Blazejewko, 2002, 120-121.
[10] B. Diveiev, Rotating machine dynamics with application of variation-analytical methods for rotors calculation. Proceedings of the XI Polish – Ukrainian Conference on "CAD in Machinery Design – Implementation and Education Problems.", Warsaw, 2003, 7-17.
[11] Z. Li, B. Diveyev, M.J. Crocker, Refined Numerical Schemes for a Stressed-Strained State of Layered Elements for damping estimation. 1, Theoretical foundations. (In press).
[12] Z. Li, B. Diveyev, M.J. Crocker. Refined Numerical Schemes for a Stressed-Strained State of Layered Elements for damping estimation. 2. Damping Bounds,Predictions and Experimental Modelling (In press).
[13] B.M. Diveyev and M.M. Nykolyshyn. Refined Numerical Schemes of a Stressed-Strained State of Structural Joints of Layered Elements, Journal of Mathematical Sciences, 107 (2001) 3666-3670.